

Kacper Kaczmarczyk. Viva Questions

1. Your liner and exhaust valve temperatures are particularly high whilst piston temperatures are low. Do you think you should have corrected these and what would be the steps to do so?

Answer: Firstly, it is important to realize that all of the component temperatures calculated by the model in the write up are that of the interior wall boundary, not the surface mean temperature, which is important since the far majority of the sources which I've looked at haven't directly measured that boundary temperature due to the inherent difficulty of such task. As such, I have expected my temperatures to be higher than what is considered to be 'expected'.

I agree that based on literature, the peak instantaneous interior wall temperatures are high, particularly for liner and exhaust valve. High exhaust valve temperatures were caused by considering the mean exhaust gas temperature as the cooling temperature, so naturally the temperature of the exhaust valve would fall somewhere between mean cylinder temperature and $\sim 700^\circ\text{C}$, as shown on page 21. Additionally, real life exhaust valve is coated with a thermal barrier, considering this, I don't think it is too

unreasonable to consider the exhaust valve temperatures to be somewhat plausible if they represent temperature of the coating. The liner temperatures were caused by iron's relative resistivity to heat transfer. Though peak instantaneous values are high, the cycle averaged liner temperatures are much more realistic. Cycle averaged piston temperatures on page 20 are below 100 °C, which is lower than expected, however, I was trying to estimate the effect the oil squirters have on the 'effective' cooling temperature. As I didn't have any real-life data it is very difficult to say whether or not my boundary temperatures are incorrect, I was only able to use the knowledge that I have gathered to estimate and simplify many variables.

The heat transfer model I have created is definitely not correct, but it is good enough to estimate the total heat transfer with reasonable confidence.

The effect the inaccuracy of the individual component interior wall temperatures has on the net heat transfer is very low and in the case of the exhaust valve, which has the smallest surface area in the engine, the effect becomes almost negligible. This is why programs such as Ricardo Wave can get away with assuming constant valve temperatures to be 550 °C.

What does matter is the heat flux, which is a parameter which can be used to compare engine to engine thermal transfer in much more dimensionless fashion. I have shown a comparison between my model's results and real-life measurements on page 17. It shows that my model has been able to capture the magnitudes and profile of heat flux very well, giving me confidence that I have achieved reasonable ballpark with my heat flux. Consequently somewhat proving the net of wall temperatures and the rest of the model to be in the ballpark as well.

Therefore, based on all of the data and research I had available, I believe that although details such as: 'at times the instantons component temperatures are not realistic', the resultant net heat transfer, especially when made relative to net heat into gas, is as good as I could make it without running my own test engine. It is important to remember that this heat transfer model is only an estimation tool and hoping it would be anything more just wasn't realistic in the context of my project.

There are various ways to improve my heat transfer model, such as creating liner zones, since liner temperature near cylinder head will be hotter, considering more heat transfer pathways, especially for the valves, creating hot and cold cylinder zones

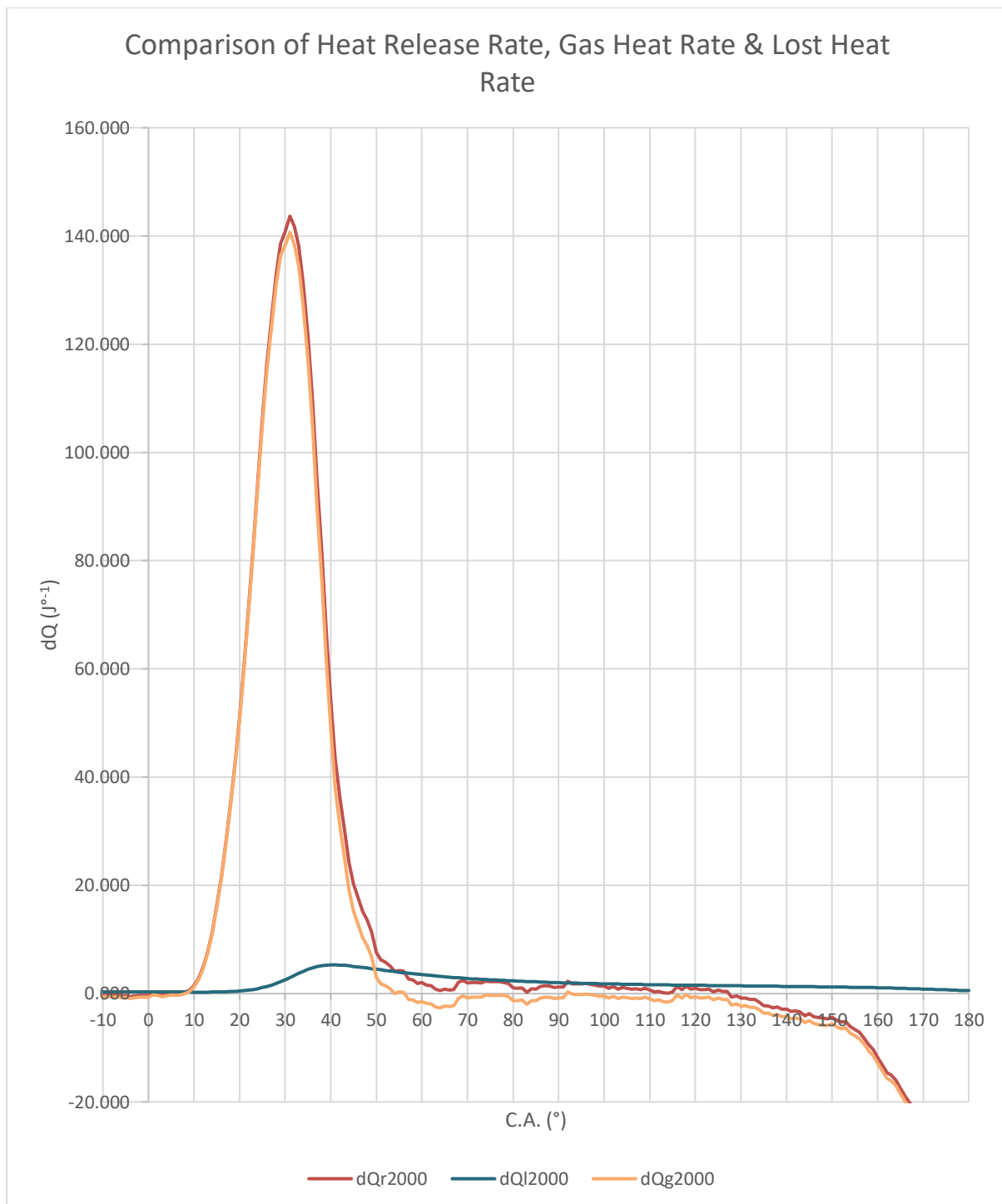
which propagate and arrive at the individual components at individual times, etc. However, without concrete data that I could use to actually define my model's accuracy, I believe such activity to be a waste of time, which was something that I couldn't afford to do at the time of working on my project. Additionally, even if my heat transfer model was 100% accurate, it still would have been altered to suite the n(s) to n(g) accuracy before and after combustion, making heat transfer model a luxury rather than necessity in my particular project.

2. In Figure 4.16 the graph shows low values for heat loss rate which do not seem to agree with the author's stated 9-16%. Can you comment and elaborate please?

Answer: Below is the table which gathers the total heat transferred through each component and calculates the total heat transferred as a % of the total released energy.

Lost Energy							
Liner (J)	125.1	131.4	114.0	86.9	75.5	64.3	54.8
Head (J)	83.3	84.4	73.6	57.9	50.5	48.4	47.9
Piston (J)	112.3	113.2	98.4	77.5	67.7	64.5	63.3
Intake Valve (J)	26.8	26.7	23.0	18.2	15.7	14.7	14.4
Exhaust Valve (J)	10.2	7.4	4.5	2.7	2.3	2.7	3.4
Total Lost Energy (J)	357.8	363.1	313.6	243.2	211.7	194.6	183.8
Total Lost Energy Fraction (%)	14.60	13.09	10.64	9.24	8.47	8.65	9.73

I believe the confusion was created because Figure 4.16 focused on a fraction of the stroke, below is the full graph:



As you can see, heat transfer spans the big majority of the expansion stroke, whereas heat into gas finishes at 55° ATDC. So even though the instantaneous values aren't high they add up to 13% of total released heat in this case.

3. In Appendix, non-corrected values for Heat Transfer Rate follow a normally expected pattern with lowest rate at lowest RPM. Why do the corrected values reverse this trend at low RPM when you apply a Heat Transfer Multiplier of less than one? With hindsight are you still happy with the corrected values and can you justify?

Answer: This trend is not reversed as such; it just does not longer apply.

As I wrote on page 36 – 37, the heat transfer correction multipliers were used to match $n(s)$ to $n(g)$ before and after combustion.

Below are the two graphs which you are concerned about:

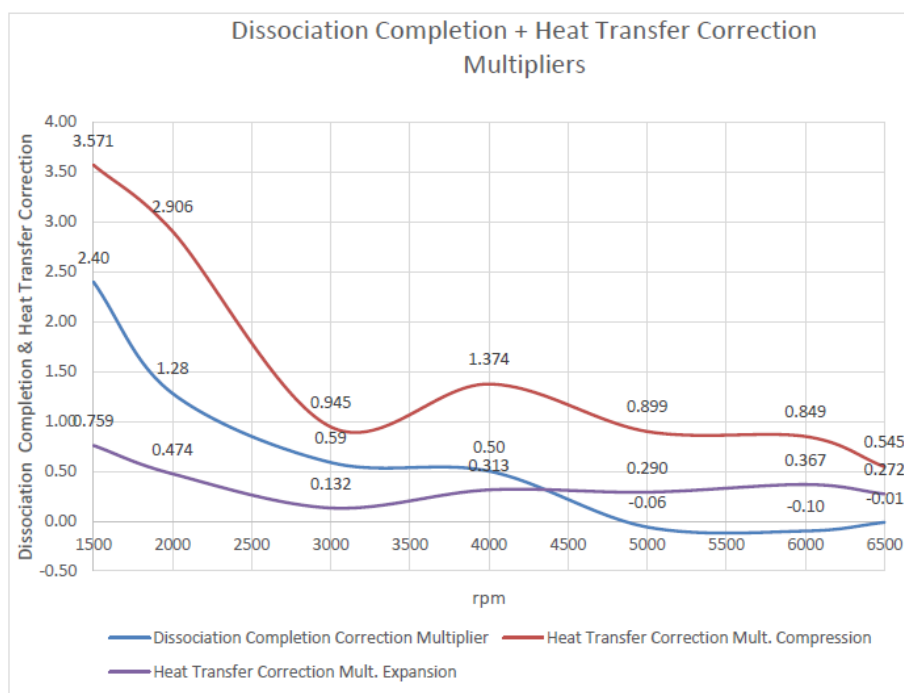


Figure 4-29: Dissociation Completion + Heat Transfer Correction Multipliers

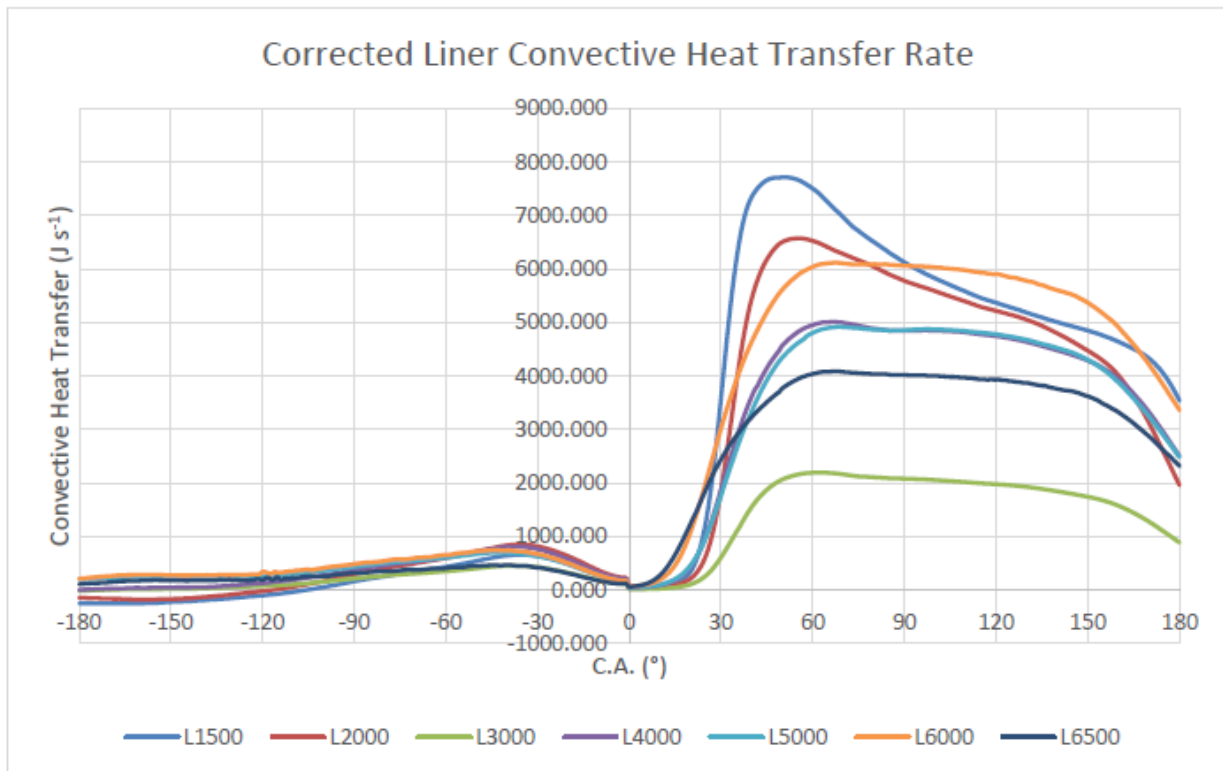


Figure 9-13: Corrected Liner Convective Heat Transfer Rate

If you look at the HTCME trend in figure 4.29, you will see that it is replicated in figure 9.13: HTCME is highest at 1,500 and 2,000rpm, so as their relative heat transfer rates, HTCME is the lowest at 3,000rpm and so as its relative heat transfer rate and after 3,000rpm the HTCME is somewhat consistent and so as their relative heat transfer rates.

As I said before, this heat transfer model was only supposed to estimate heat transfer rate and I have used it specifically in my project to fit $n(s)$ and $n(g)$ relation before and after combustion. Though the absolute heat transfer values may indeed seem incorrect, this doesn't matter as the entire point was to satisfy my $n(s)$ to $n(g)$ assumption, which would have

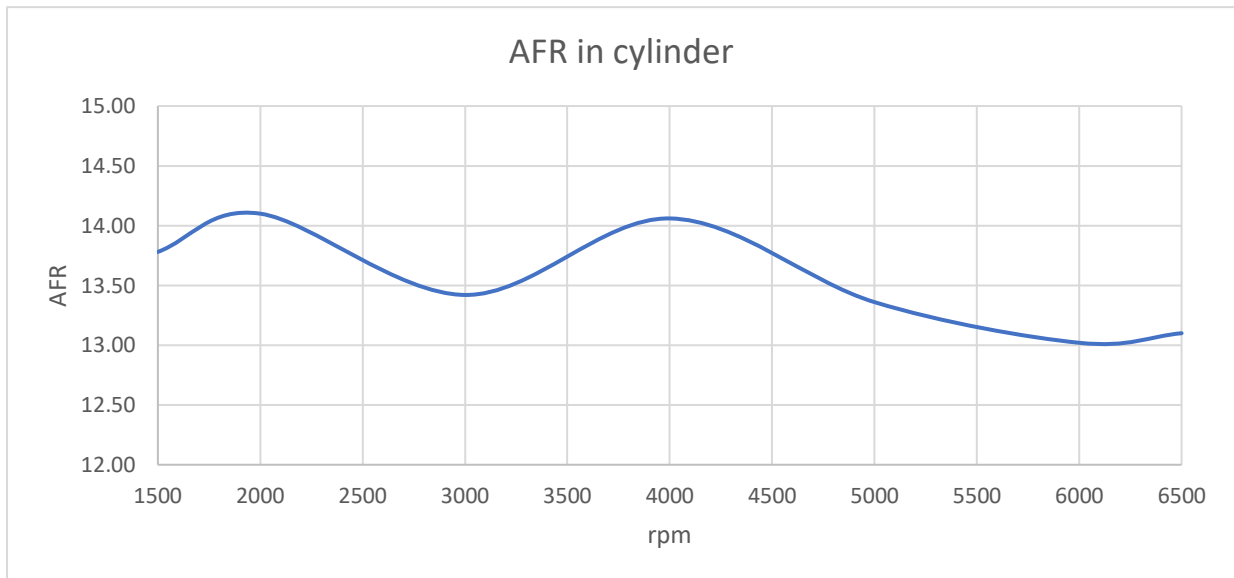
caused very similar effect even on a perfect heat transfer model.

I do believe that the discrepancy in heat transfer model suggests an issue exists somewhere in the simulation. However, the entire reason behind the 3 correction factors was to correct any possible inaccuracies in order to ensure upmost accuracy of the $n(s)$ values. The project wasn't about creating accurate heat transfer model, it was about finding out what the real polytropic index during combustion is and in order to calculate it I had to focus solely on $n(s)$ accuracy and abandon the desire to create a perfect ICE model from 1st principles, in order to make the project feasible. I did mention this in the conclusion.

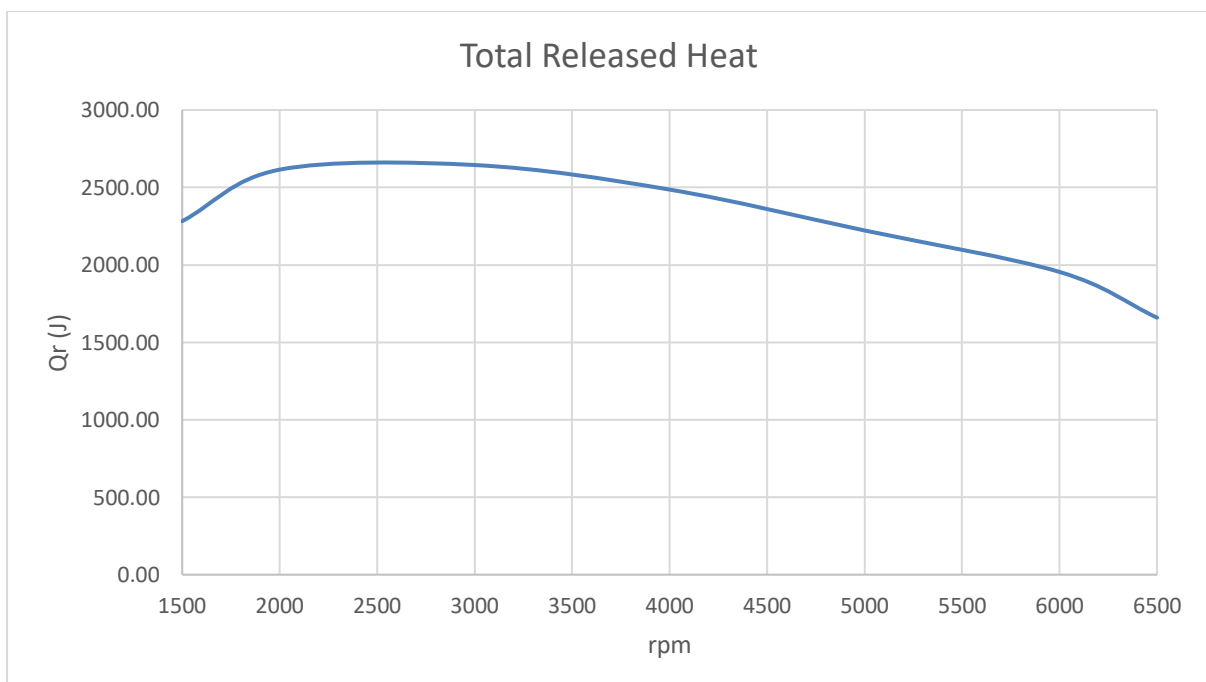
4. In Appendix, do you believe that burned temperatures are highest at low RPM as shown in Figure 9.33? Nominal exhaust AFR is highest at this point and unburned temperatures are lowest at this point as shown in Figure 9.30. Please discuss.

Answer: I haven't used the exhaust AFR as I believe the in cylinder AFR to be more reasonable. In my model AFR at low rpm isn't highest, as shown on the graph

below:



I do not believe the burnt temperature at low rpm to be highest. They do have the highest peak values, but those burnt temperatures fall the quickest, not to mention their low duration of high temperature. On average during combustion they all seem to have similar values, though I never calculated this.



Looking at the graph above I would expect the highest mean burnt temperatures during combustion to occur at 2,000 and 3,000rpm. Though naturally due to the way I calculate it the MFB profile has the biggest influence on these values, especially the peaks.

The trend of un-burnt temperatures is mainly caused by the adiabatic assumption and the mean cylinder temperatures.

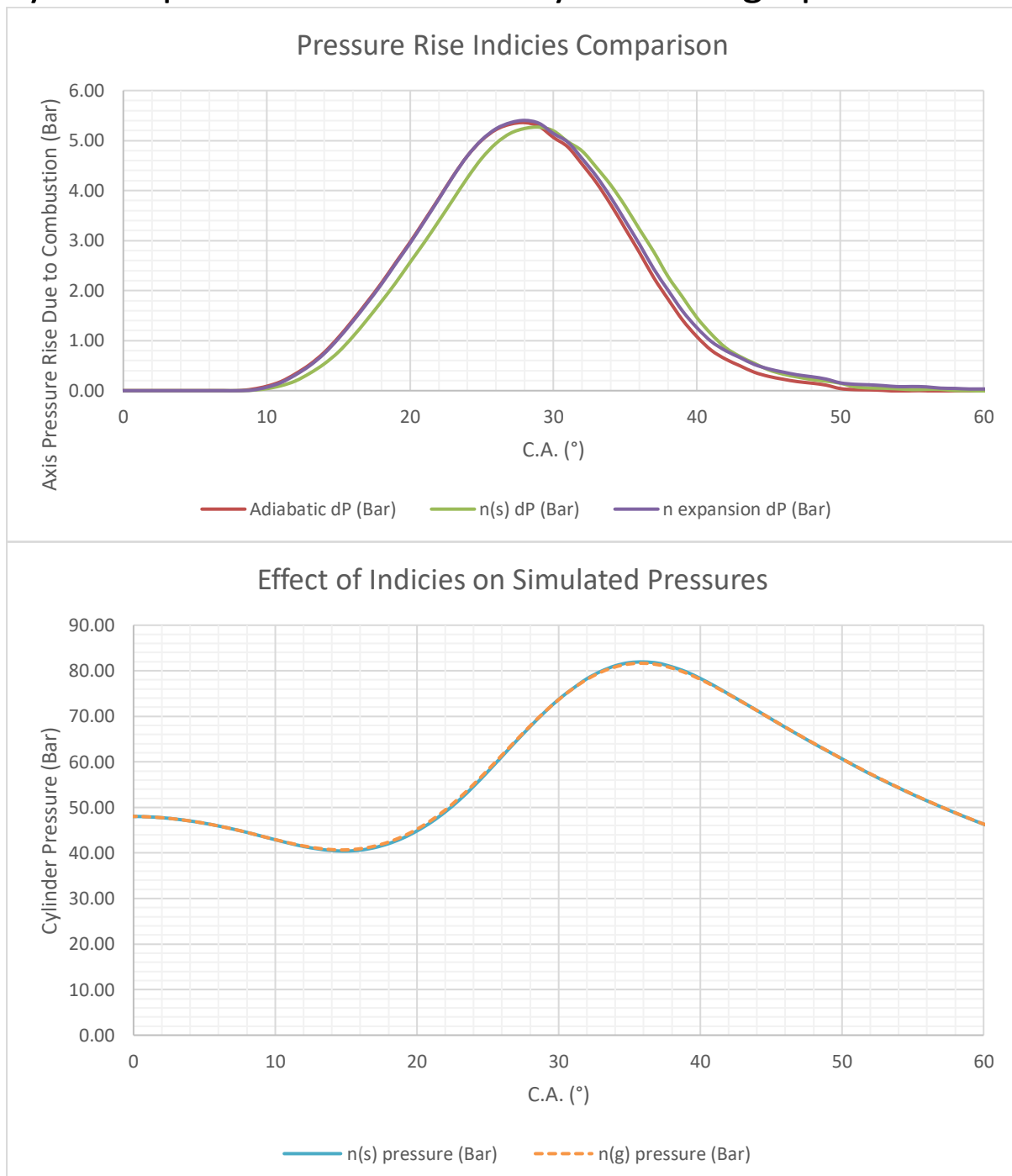
5. A) You comment that $n(s)$ is an abstract and that $n(s)$ is proportional to $dn(g)$. Can you elaborate?
B) Why do you go on to say it could be replaced by an adiabatic process and could be replaced satisfactorily by n in expansion? C) Can you explain why at 6500RPM in Figure 9.28 $n(s)$ does not show good correlation with $dn(g)$?

Answer:

A) $n(s)$ is abstract since it is not a measurable/ physical property, it is hidden inside $n(g)$ making it difficult to imagine what $n(s)$ looks like.

According to my findings, a good relationship exists between $n(s)$ and $dn(g)$, making $n(s)$ even more abstract if the results are to be believed, as I personally find it difficult to imagine it in physical context.

B) As I wrote on page 43, the resultant $n(s)$ values have a low effect on the simulated pressures when utilising classical 0D techniques. Below is a graph which shows that although a difference exists, primarily before peak pressure increase, the difference is relatively small (max of $\sim 10\%$), especially when made relative to the cylinder pressures as shown by the next graph.



C) I assume that this is the graph you meant:

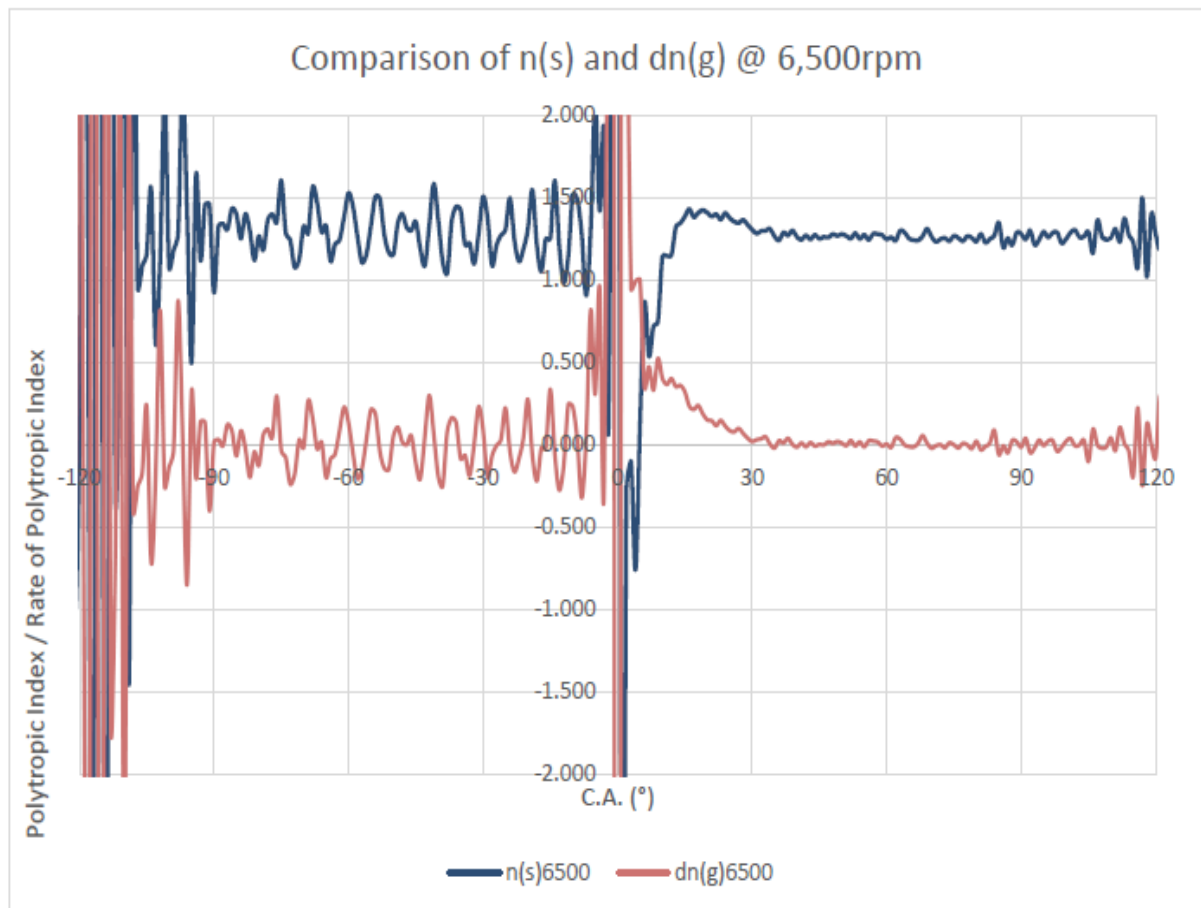


Figure 9-58: Comparison of $n(s)$ and $dn(g)$ @ 6,500rpm

Before and after combustion $n(s)$ still resembles $dn(g)$ very well. However, it doesn't relate well during combustion until the very end.

I believe this has been caused by the reversal of dV at TDC. The same effect can be seen at 6,000rpm during the few initial C.A. Though, SOC at 6,500rpm occurs earlier so TDC occurs at more significant point in the $dn(g)$ profile so it requires more C.A. to normalize.

Increasing resolution (decreasing the $dC.A.$) would help this tremendously.